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JP 050080561 B JP 030055643 B JP 020135604 U

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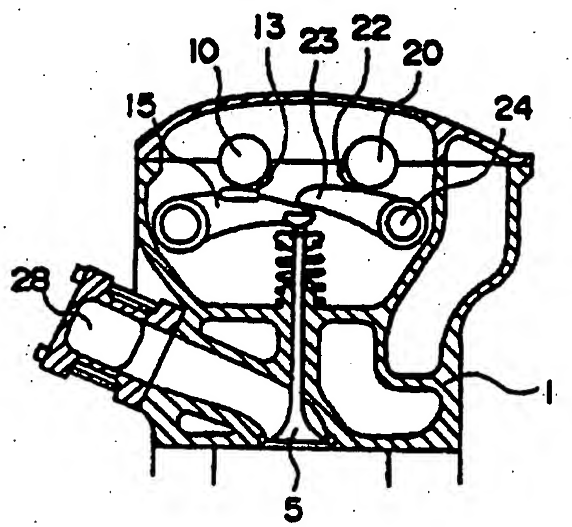
(54) **Variable compression ratio engine**

(57) A variable compression ratio engine which can operate both in a Miller cycle and a normal cycle and which can produce a high output, reduce the generation of NOx and prevent the occurrence of knocking. In order to make this happen, the engine comprises an exhaust gas recirculating device comprising in turn a first cam shaft (10) provided with cams (11, 12, 13) for operating an intake valve (2) and exhaust valves (4, 5) and a second cam shaft (20) provided with cams (21, 22) for operating at least an intake valve (3) and an exhaust valve (5) to thereby recirculate a part of exhaust gas into intake gas.



GB 2 301 398 A

FIG. 1

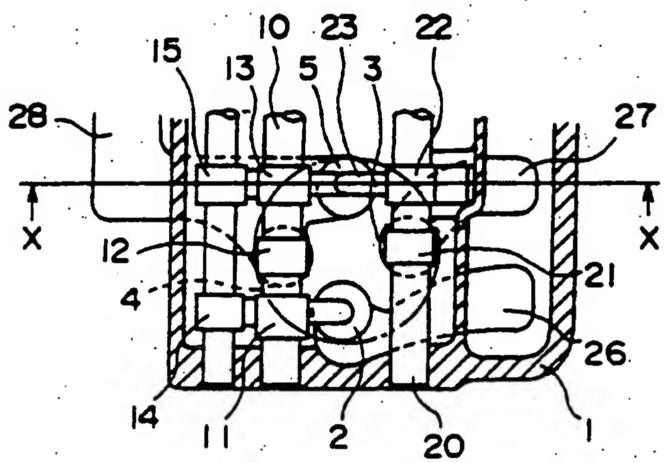
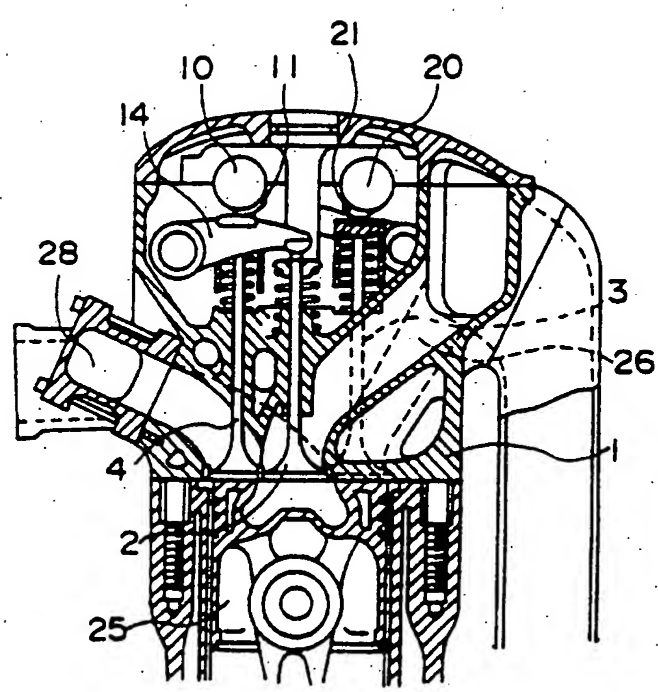

10
23
22
15
13
5
3
28
27
X
X
12
21
4
26
14
11
2
20
1

FIG. 2


21
10
11
20
14
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3
26
4
1
2
25

FIG. 3

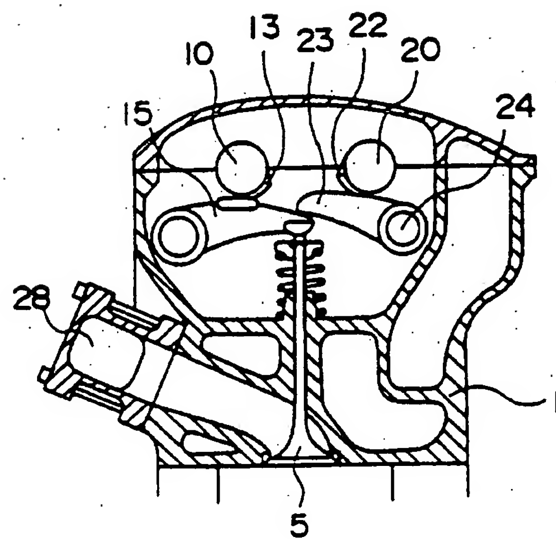


FIG. 4

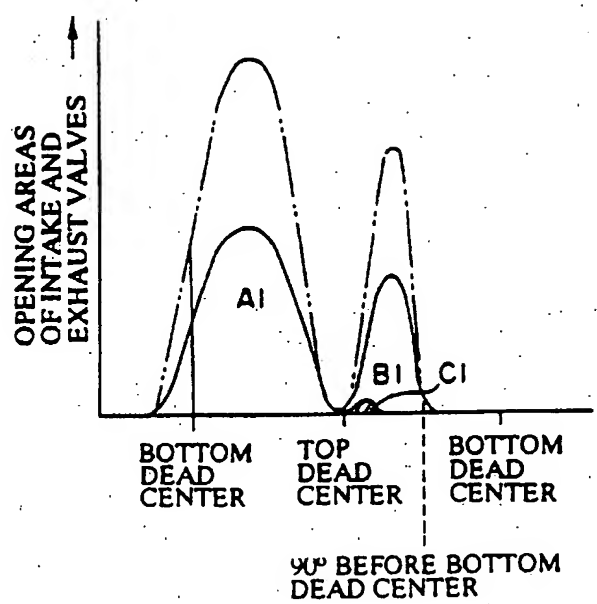

OPENING AREAS OF INTAKE AND EXHAUST VALVES
A1
B1
C1
BOTTOM DEAD CENTER
TOP DEAD CENTER
BOTTOM DEAD CENTER
90° BEFORE BOTTOM DEAD CENTER

FIG. 5

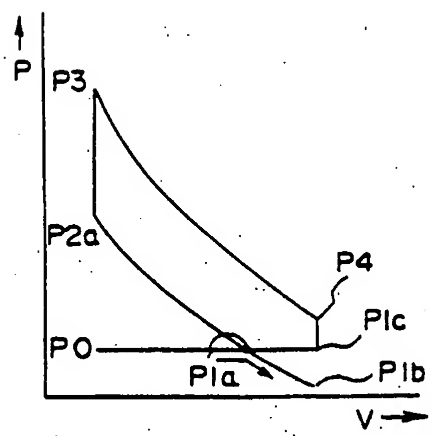

P
P3
P2a
P4
P1c
P0
P1a
P1b
V

FIG. 6

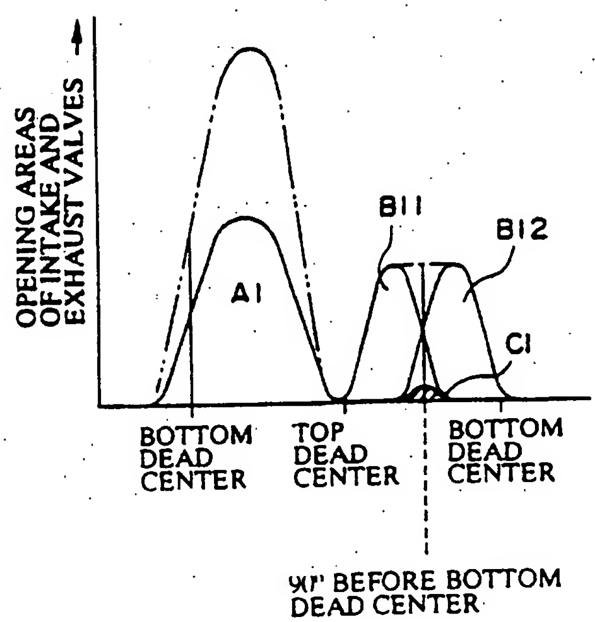

OPENING AREAS OF INTAKE AND EXHAUST VALVES
A1
B11
B12
C1
BOTTOM DEAD CENTER
TOP DEAD CENTER
BOTTOM DEAD CENTER
90° BEFORE BOTTOM DEAD CENTER

FIG. 7

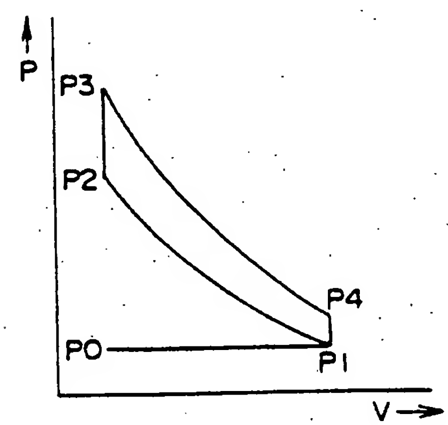

P
P3
P2
P4
P0
P1
V

FIG . 8

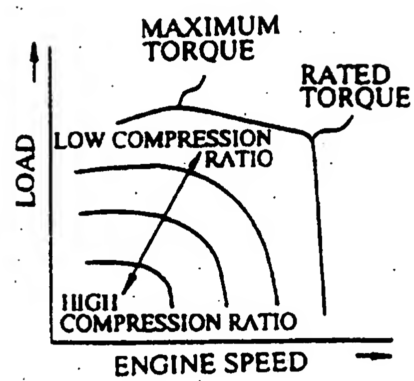

MAXIMUM
TORQUE
RATED
TORQUE
LOW COMPRESSION
RATIO
LOAD
HIGH
COMPRESSION RATIO
ENGINE SPEED

FIG . 9

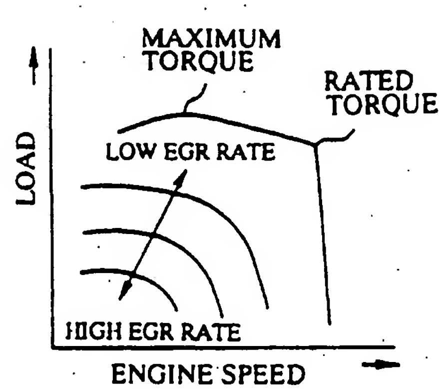

MAXIMUM
TORQUE
RATED
TORQUE
LOW EGR RATE
LOAD
HIGH EGR RATE
ENGINE SPEED

FIG. 10

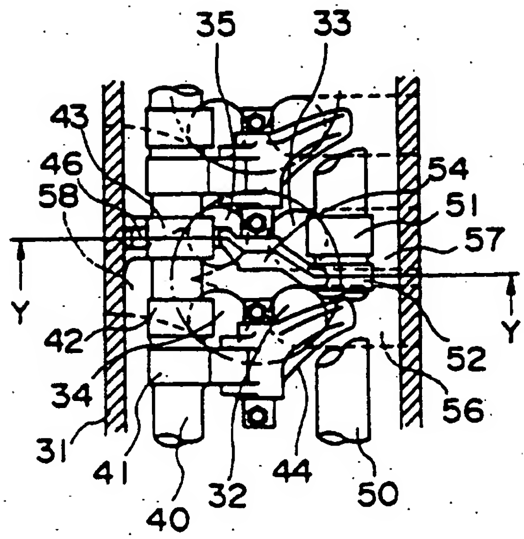
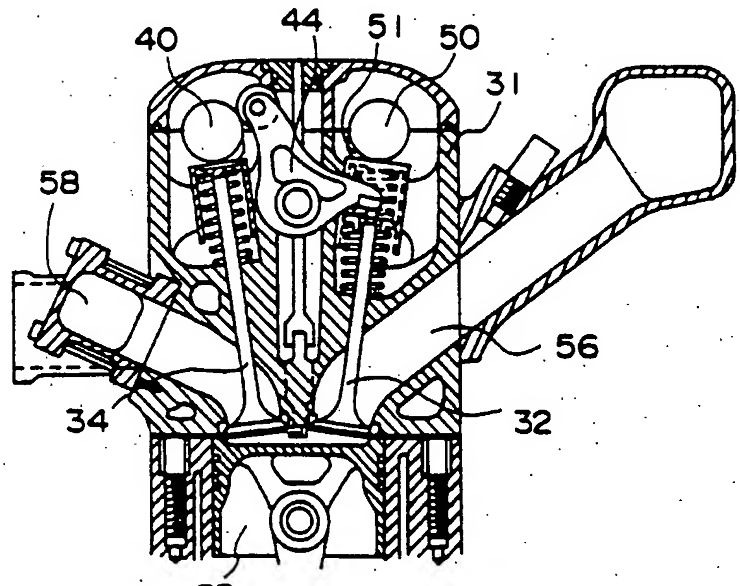

35
33
43
46
58
54
51
57
Y
Y
42
52
34
56
31
41
32
44
50
40

FIG. 11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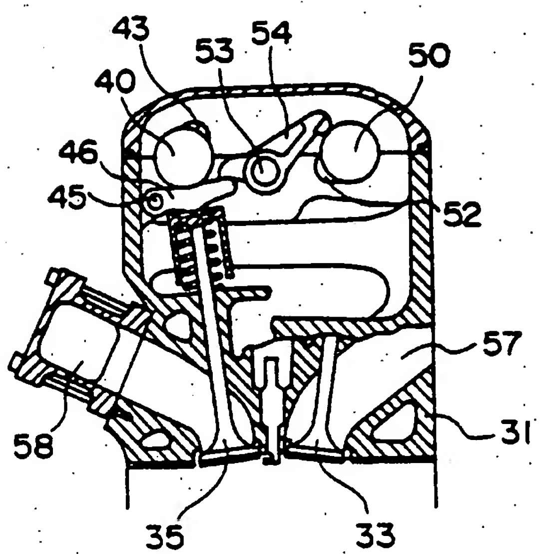
44
51
50
31
58
56
34
32
55

FIG. 12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45
52
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31
58
35
33

FIG. 13

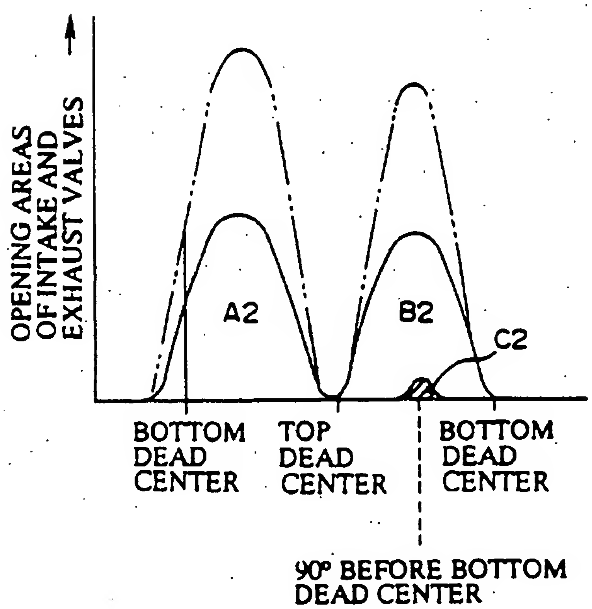

OPENING AREAS OF INTAKE AND EXHAUST VALVES
A2
B2
C2
BOTTOM DEAD CENTER
TOP DEAD CENTER
BOTTOM DEAD CENTER
90° BEFORE BOTTOM DEAD CENTER

FIG.14

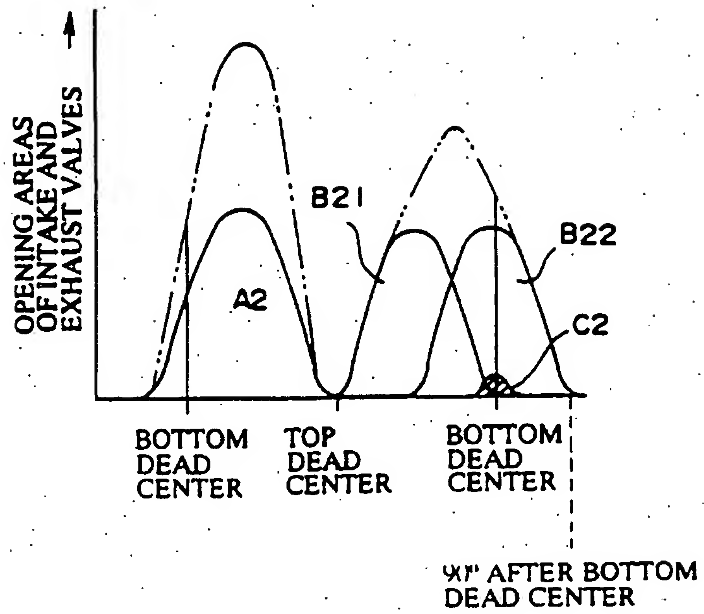

OPENING AREAS OF INTAKE AND EXHAUST VALVES
B21
B22
A2
C2
BOTTOM DEAD CENTER
TOP DEAD CENTER
BOTTOM DEAD CENTER
AFTER BOTTOM DEAD CENTER

FIG.15

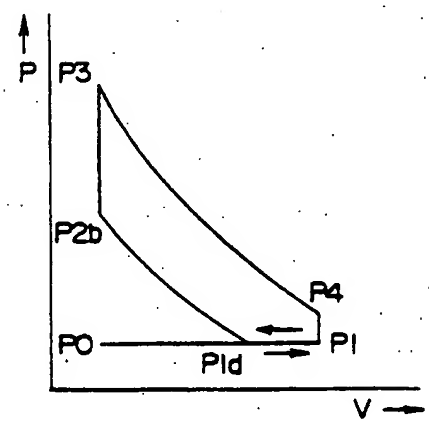

P
P3
P2b
P4
P0
P1
P1d
V

FIG. 16

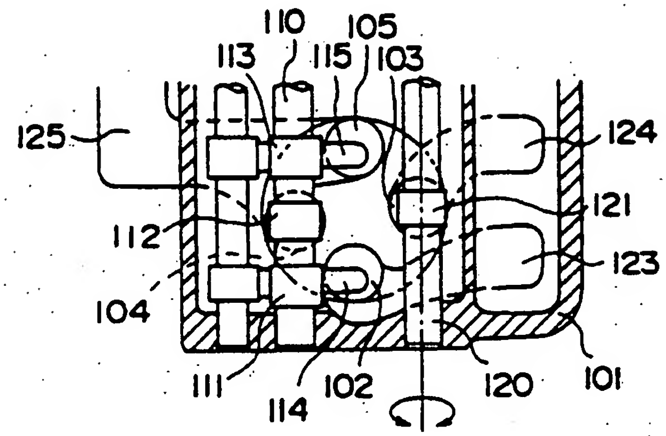

110
105
113
115
103
125
124
121
112
123
104
111
114
102
120
101

FIG. 17

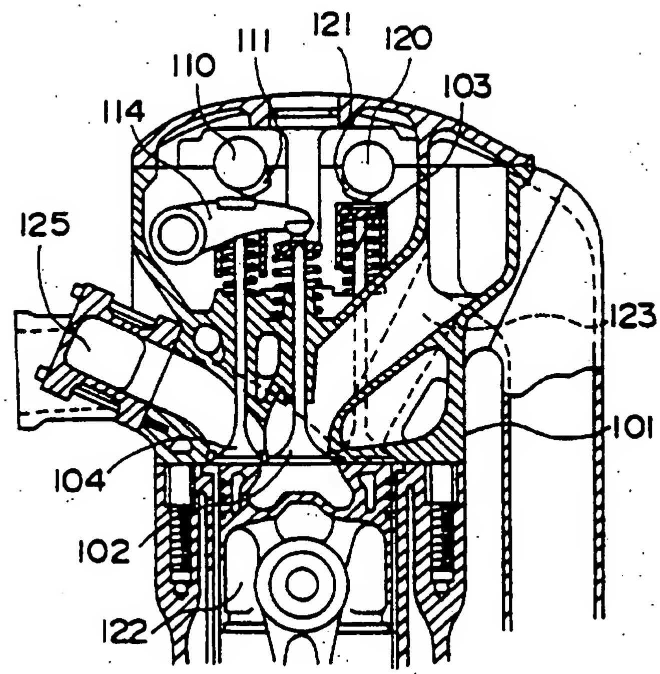

121
111
120
110
103
114
125
123
101
104
102
122

FIG. 18

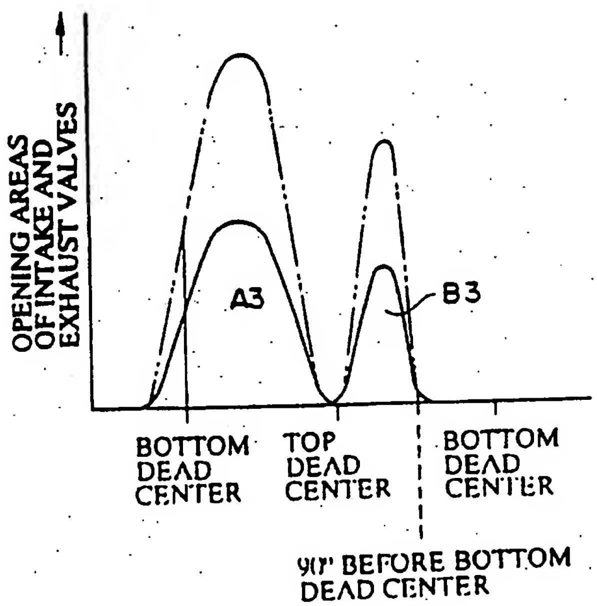

OPENING AREAS
OF INTAKE AND
EXHAUST VALVES
A3
B3
BOTTOM
DEAD
CENTER
TOP
DEAD
CENTER
BOTTOM
DEAD
CENTER
90° BEFORE BOTTOM
DEAD CENTER

FIG. 19

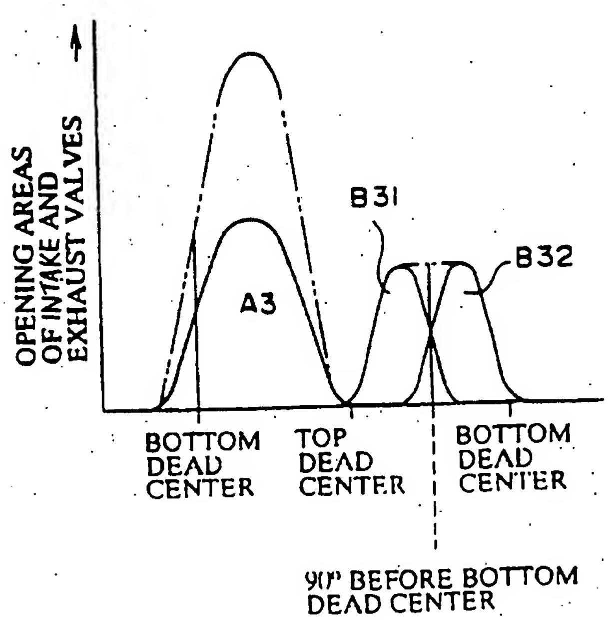

OPENING AREAS
OF INTAKE AND
EXHAUST VALVES
B31
B32
A3
BOTTOM
DEAD
CENTER
TOP
DEAD
CENTER
BOTTOM
DEAD
CENTER
90° BEFORE BOTTOM
DEAD CENTER

FIG.20A

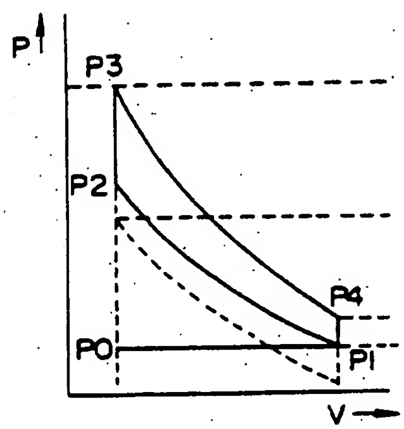


FIG.20B

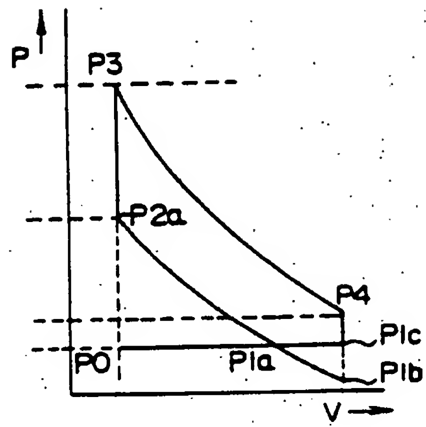


FIG.21

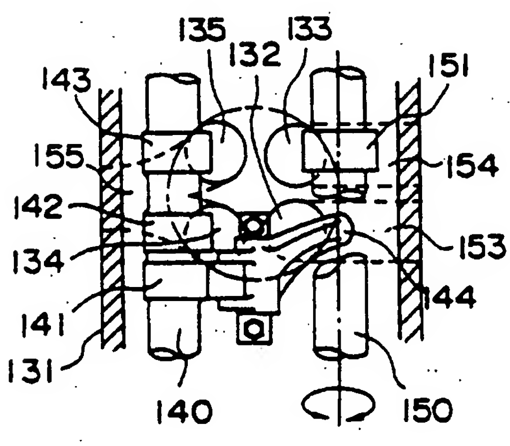


FIG. 22

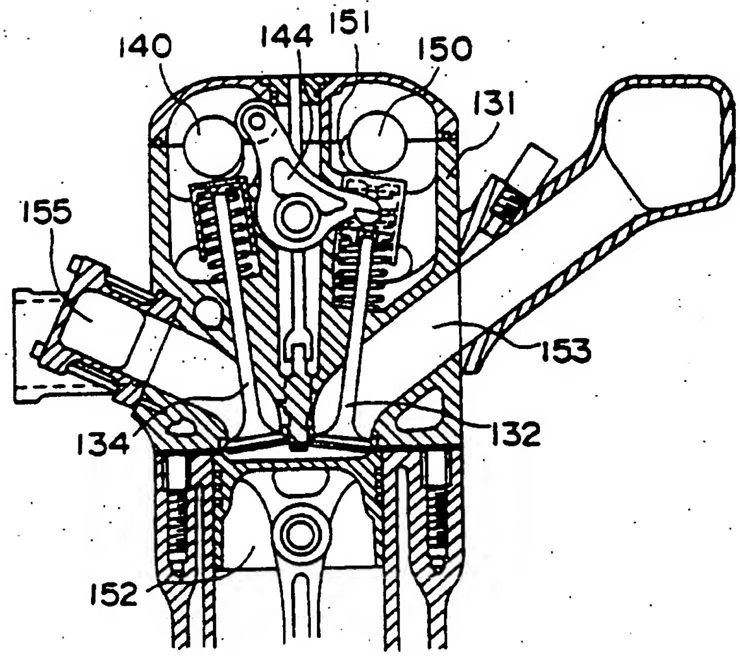

140
144
151
150
131
155
153
134
132
152

FIG. 23

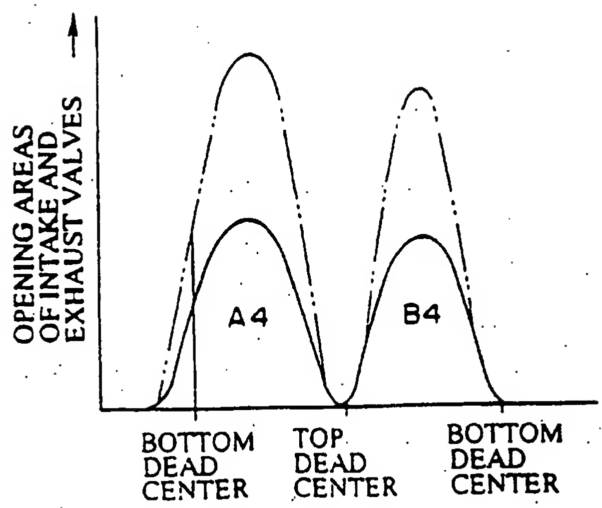

OPENING AREAS OF INTAKE AND EXHAUST VALVES
A4
B4
BOTTOM DEAD CENTER
TOP DEAD CENTER
BOTTOM DEAD CENTER

FIG. 24

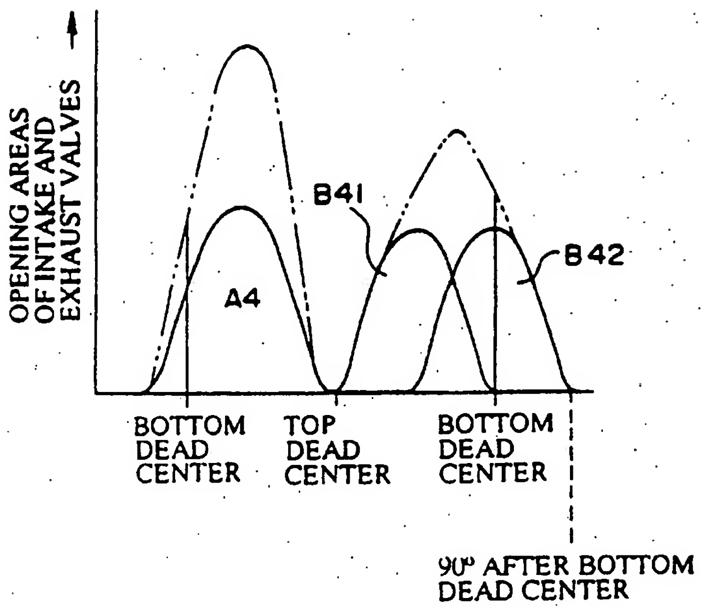

OPENING AREAS OF INTAKE AND EXHAUST VALVES
B41
B42
A4
BOTTOM DEAD CENTER
TOP DEAD CENTER
BOTTOM DEAD CENTER
90° AFTER BOTTOM DEAD CENTER

FIG. 25

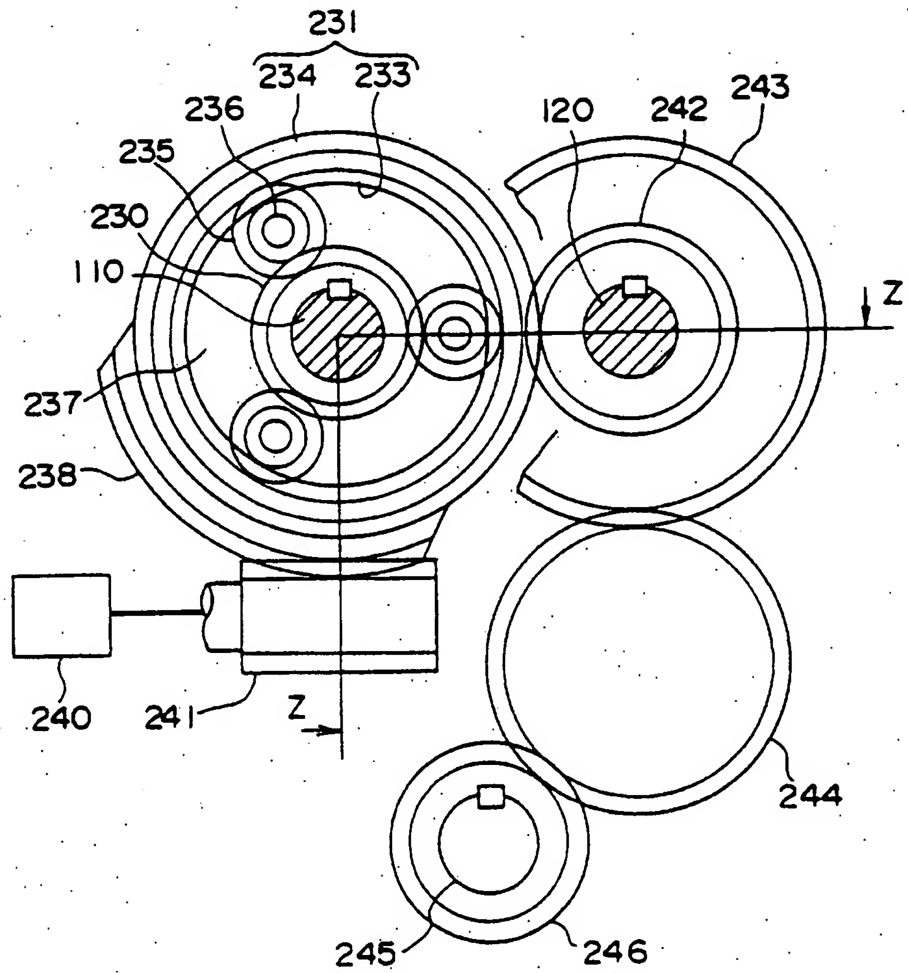

231
234
233
236
235
230
110
237
238
120
242
243
Z
240
241
Z
244
245
246

FIG. 26

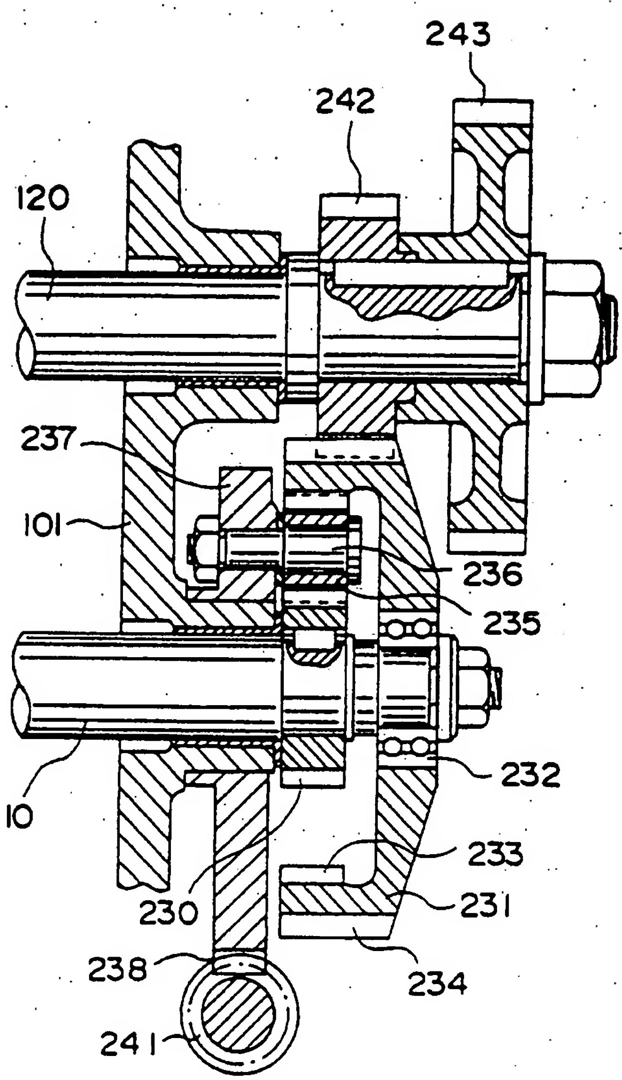


FIG. 27 CONVENTIONAL ART

COMPRESSION RATIO	17			12		
AVERAGE AXIAL EFFECTIVE PRESSURE	13	21	34	13	21	34
BOOST PRESSURE	2	3.1	5	2	3.1	5
COMPRESSION PRESSURE	97	150	242	60	93	150
EXPLOSION DEGREE	1	1	1	1	1	1
CYLINDER PRESSURE Pmax	97	150	242	60	93	150

FIG. 28 CONVENTIONAL ART

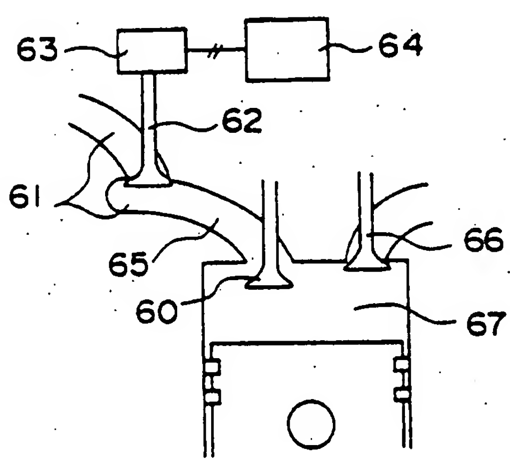


FIG.29A CONVENTIONAL ART

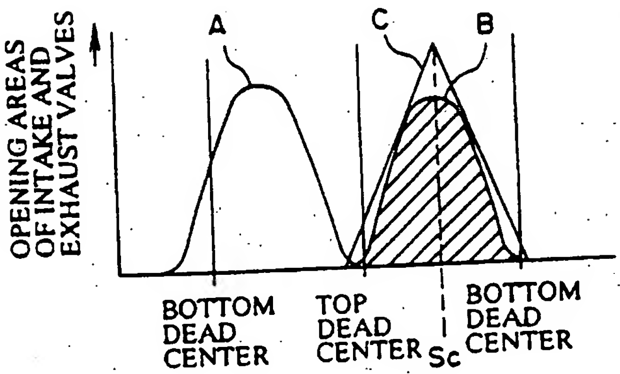
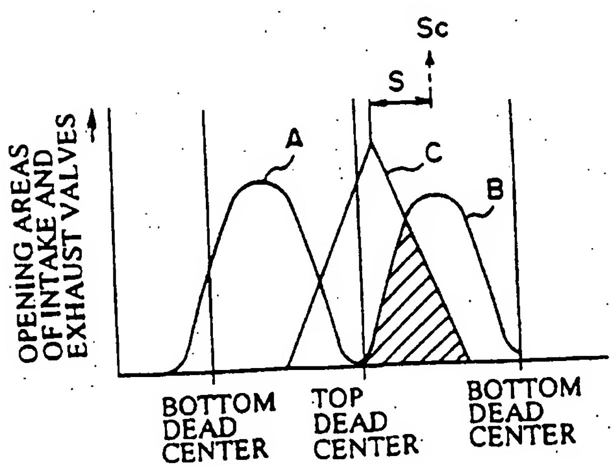


FIG.29B CONVENTIONAL ART



VARIABLE COMPRESSION RATIO ENGINE

The present invention relates to a variable compression ratio engine, and more particularly, to a variable compression ratio engine which can convert between an ordinary cycle and a Miller cycle.

In order to reduce NOx contained in exhaust gas, exhaust gas recirculation (EGR), by which exhaust gas which is an inert gas is recirculated into an intake gas and by which the combustion temperature is lowered, has been conventionally conducted in vehicle engines. Regarding this exhaust gas recirculation, when the load on an engine is heavy, the volume efficiency is improved as the temperature of EGR gas becomes lower, and the combustion temperature becomes lower and NOx decreases as there is a larger amount of EGR gas. On the other hand, when the load on an engine is light, thecombustion is not stable when the temperature of EGR gas is low, so that EGR gas with high temperature is preferable. For this reason, a method of controlling the EGR gas to be cooled when the load is heavy and of controlling the EGR gas so that the EGR gas is not cooled when the load is light by providing an EGR gas cooling means is already known (for example, refer to Japanese Laid-Open Patent Application No. 4175453 and Japanese Laid-Open Patent Application No. 4301172).

However, when the EGR is conducted with a heavy load, disadvantages of increased fuel consumption, reduced output and so on are brought about.

In other conventional art, many of the compression ratios of engines, for example, direct injection type Diesel engines are set in the vicinity of 15 to 17. This compression ratio is required for securing starting efficiency and a good combustion state when a load is light, for example, a combustion state without blue and white smoke including hydrocarbon compounds etc. The times for opening and closing an intake valve are fixed in order that the aforementioned compression ratio is obtained. When the compression ratio is determined, the pressure within a cylinder at the end of the engine's compression stroke is determined, and the pressures within the cylinder at ignition, explosion, and so on are also determined. Meanwhile, a maximum allowable pressure within a

cylinder is determined in accordance with an engine, and the higher the compression ratio is, the higher the pressure within a cylinder becomes at the end of compression. Accordingly, the difference between this pressure within a cylinder and the maximum allowable pressure within a cylinder decreases, and this is the main reason which prevents the engine from achieving a high power output.

The aforementioned compression ratio is desired to be in the vicinity of 11 to 13 from the viewpoint of combustion efficiency and high power output. As an example, average axial effective pressures which can be achieved with the compression ratios at 17 and 12 are shown in Figure 27. A unit of pressure in Figure 27 is kgf/cm^2. For example, in the case of an engine with maximum allowable pressure within the cylinder Pmax is 150kgf/cm^2 or more (Pmax $\leq$ 150kgf/cm^2), the average axial effective pressure stays at 21kgf/cm^2 with the compression ratio of 17, but with the compression ratio of 12, the average axial effective pressure can be 34kgf/cm^2, that is, a high power output can be achieved.

However, since it is an absolutely necessary requirement that excellent starting and excellent combustion state are obtained when the load is light, in the current state of the art the compression ratio is set in the vicinity of 15 to 17 and high power output is sacrificed. This is also the case in gasoline engines, and though the compression ratio is desired to be 11 to 13 as in Diesel engines from the view of combustion efficiency (thermal efficiency), the compression ratio is set at 8 to 10 in order to prevent the occurrence of knocking when the load is heavy. As a result, there are disadvantages of increasing fuel consumption and of generating a large amount of CO_2.

As the art which improves thermal efficiency of Diesel engines and which reduces exhaust emission, a Miller cycle engine by which a low compression ratio and high expansion ratio can be obtained has been known. The Miller cycle engine has two types; a type which blocks the flow of intake gas in the middle of an intake stroke by closing an intake valve at an early stage, and a type which lets intake pressure escape at the beginning of a compression stroke by closing the intake valve at a later stage. However, as described above, when a Miller cycle engine is

operated at a low speed and light load range of the engine, the effective compression ratio is reduced and therefore there is a risk of unstable ignition.

As a means of eliminating this disadvantage, there is the Miller cycle engine described below (refer to, for example, Japanese Laid-Open Patent Application No. 63-277815). In Figure 28, an intake valve 60 is opened and closed by means of a crankshaft, timing gear, cam shaft, tappet, push rod, and rocker arm which are not illustrated in the drawing. At the middle of an upstream passage 61 of the intake valve 60, a new valve 62 is provided, and the engine speed, a load and so on are detected as signals. Based on this detection, the valve 62 is closed earlier than the intake valve 60 is closed by a valve mechanism 63 connecting to a conversion mechanism 64 according to the driving conditions, in other words, an early closing Miller cycle operation is conducted. Reference number 66 refers to an exhaust valve, and 67 to a cylinder chamber. The valve 62 and the valve mechanism 63 may be rotary valves.

Figures 29A and 29B show the relationship between the position of a piston (abscissa) of the aforementioned engine and the area of the opening, and a curve A corresponds to the exhaust valve 66, a curve B corresponds to the intake valve 60, and C shown by two lines corresponds to the valve 62. As shown in Figure 29A, when the load is light, the intake valve 60 and the valve 62 open and close at the same time, so that the area of the opening of the intake valve 60 becomes a hatched portion, and the engine operates in a normal cycle. On the other hand, when the load is heavy, the valve 62 opens and closes earlier by as much as S, as Figure 29B shows, the actual opening area of the intake valve 60 becomes the hatched portion. Accordingly, the intake valve 60 closes early with the actual compression ratio being low, so that the engine operates in an early closing Miller cycle, and a high power output can be achieved.

However, even if the valve 62 is closed by operating the engine in a Miller cycle as described above, the volume of air in a passage 65 between the intake valve 60 and the valve 62 is added to the volume of air in a cylinder 67 while the intake valve 60 opens. Accordingly the volume

increases, so that the effect of closing the valve 62 in the middle of the intake stroke is decreased and the effect of the Miller cycle is reduced. There is a disadvantage of a pumping loss caused by an increase of intake resistance immediately before the valve 62 closes and by the comings and goings of intake gas resulting from the amount of air in the passage 65 becoming an excessive volume.

Further, in each type of engine, it is an important feature to make the opening and closing times of the valves variable in order to obtain high torque generated over a large range of rotational speeds. For example, as a practical method of making a valve timing variable, a method of changing a phase of a timing gear and a cam shaft by engaging the cam shaft with the timing gear by means of a helical spline, and by moving the timing gear in an axial direction relative to the cam shaft, has been known (refer to, for example, Japanese Laid-Open Patent Application No. 61-85515).

However, although high torque can be obtained over a wide speed range in the aforementioned structure, the angle variations of the cam shaft can not normally be made to be 20° to 40° or more at a given crank angle, so that it is difficult to make a helical angle of the helical spline extremely large. Accordingly, in order to convert between a normal cycle (an Otto-cycle, a Diesel cycle and so on) and a Miller cycle by changing the opening and closing times of the valves in order to output high power, the angle variations of the cam shaft needs to be 70° to 90° at the given crank angle and the conventional helical spline type is insufficient.

In order to eliminate the aforementioned disadvantages of the conventional art, an object of the present invention is to provide a variable compression ratio engine which can convert between a normal cycle and a Miller cycle and which has a sufficient effect of a Miller cycle. Another object is to always conduct a most suitable EGR in a wide driving range of an engine by increasing the EGR rate (the amount of EGR gas supply) under a light load and by decreasing the EGR rate under a heavy load.

The first aspect of the variable compression ratio engine relating to the present invention includes an exhaust gas recirculating device equipped with a first cam shaft provided with cams for operating an intake valve and exhaust valves and a second cam shaft provided with cams for operating at least one intake valve and an exhaust valve, and by including a valve driving device by which the closing time of the intake valve is set at a time before a piston is at the bottom dead centre with the opening and closing times of the exhaust valve at a time when the piston is in the vicinity of the top dead centre in the intake stroke at a specified driving time, and by which, in the intake stroke under a light load, the closing time of at least one intake valve is set at a time when the piston is in the vicinity of the bottom dead centre with the opening and closing times of the exhaust valve being set at a time before the piston is at the bottom dead centre, to thereby recirculate part of the exhaust gas into intake gas when a load is light by changing the phase of the cams on the second cam shaft by the valve driving device, in the variable compression ratio engine which is provided with two or more intake valves and one or more exhaust valves per cylinder and which changes the compression ratio by opening and closing the intake valves and/or the exhaust valves by the cams provided on two or more cam shafts (the present structure shall be the first setting of the opening and closing times of the valves).

The aforementioned valve driving device may be a valve driving device by which, in the intake stroke under a light load, the closing time of the intake valves is set at a time when the piston is in the vicinity of the bottom dead centre with the opening and closing times of the exhaust valve being set at a time before the piston is at the bottom dead centre, and by which the closing time of at least one intake valve is set at a time after the piston is at the bottom dead centre with the opening and closing times of the exhaust valve being set at a time when the piston is in the vicinity of the bottom dead centre, and this valve driving device may recirculate part of the exhaust gas into intake gas under a light load by changing the phase of the cams on the second cam shaft (the present structure shall be the second setting of the opening and closing times of the valves).

By the aforementioned structure, in the case of the first setting of the opening and closing times of the valves, under a heavy load, the engine operates in an early closing Miller cycle with a low compression ratio so that the exhaust gas recirculation is hardly conducted, and under a light load, the engine operates in a normal cycle with a high compression ratio with significant exhaust gas recirculation. In the case of the second setting of the opening and closing times of the valves, under a light load, the engine operates in a normal cycle with a high compression ratio and significant exhaust gas recirculation, and under a heavy load the engine operates in a late closing Miller cycle with a low compression ratio and virtually no exhaust gas recirculation.

According to a second aspect, the variable compression ratio engine includes an intake device which changes the valve timing of at least one of the intake valves by changing the phase of the cam for opening and closing the intake valve, to thereby set the closing time of the intake valve at a time before the piston is at the bottom dead centre, and to thereby set, under specified driving conditions, the closing time of at least one of the intake valves at a time when the piston is in the vicinity of the bottom dead centre. The closing time of the intake valves which is set at the time before the piston is at the bottom dead centre may be at a time when a crank rotational angle is at 20° to 90° before the piston is at the bottom dead centre.

By the aforementioned structure, under specified driving conditions, for example, at starting or under a light load, the compression ratio can be increased, so that an excellent start or combustion state can be secured. When the closing time of the intake valves is set at 20° to 90° before the piston is at the bottom dead centre, the compression ratio can be reduced, so that the pressure within a cylinder at the end of the compression is lowered. Accordingly, a margin up to the maximum allowable pressure is made, so that high power can be outputted.

According to a third aspect, of the variable compression ratio engine includes an exhaust gas recirculating device equipped with the first cam shaft provided with the cams operating the intake valve and the

exhaust valves and the second cam shaft provided with the cams for operating at least one of the intake valves and the exhaust valve, a planetary gear unit equipped with a sun gear, a ring gear which is fixedly attached at the first cam shaft, a gear which is meshed with this ring gear and fixedly attached at the second cam shaft, and a planet gear, and a variable valve timing device which changes the valve timing by adjusting the phases of the first and the second cam shafts by freely changing a relative positional relationship between the support shaft of the planet gear and the shaft of the sun gear, so that part of the exhaust gas is recirculated into intake gas by operating the variable valve timing device.

By the aforementioned structure, the relative positional relationship between the support shaft of the planet gear and the shaft of the sun gear can be changed to be a different positional relationship by operating the variable valve timing device. Accordingly, the phase of one cam shaft can be changed relative to the other cam shaft, so that the exhaust gas recirculating device can be operated. By this mechanical structure, exhaust gas recirculation can be conducted when needed as in the first aspect of the invention.

Embodiments of the invention will now be described in detail, with reference to the accompanying drawings, in which:

Figure 1 is a transverse cross sectional view of the cylinder head portion of a Diesel engine provided with an exhaust gas recirculating device according to a first embodiment of the present invention;

Figure 2 is a longitudinal sectional view of the engine in Figure 1;

Figure 3 is a sectional view along line X to X of Figure 1 and a longitudinal sectional view of the exhaust gas recirculating device of the engine;

Figure 4 is a graph showing the relationship between the movement of the piston and the opening areas of the intake and exhaust valves under a heavy load on the engine relating to the first embodiment;

Figure 5 is a PV diagram under a heavy load on the engine relating to the first embodiment;

Figure 6 is a graph showing the relationship between the movement of the piston and the opening areas of the intake and exhaust valves under a light load on the engine relating to the first embodiment;

Figure 7 is a PV diagram under a light load on the engine relating to the first embodiment;

Figure 8 is a graph showing the relationship between the variation of the compression ratio and the load on the engine relating to the first embodiment;

Figure 9 is a graph showing the relationship between the variation of the EGR rate and the load on the engine relating to the fist embodiment;

Figure 10 is a transverse cross sectional view of the cylinder head of a gasoline engine provided with the exhaust gas recirculating device relating to the second embodiment of the present invention;

Figure 11 is a longitudinal sectional view of the engine in Figure 10;

Figure 12 is a section along line Y to Y of Figure 10 and a longitudinal sectional view of the exhaust valve driving device of the engine;

Figure 13 is a graph showing the relationship between the movement of the piston and the opening areas of the intake and exhaust valves when a light load is on the engine of the second embodiment;

Figure 14 is a graph showing the relationship between the movement of the piston and the opening areas of the intake and exhaust valves when a heavy load is on the engine of the second embodiment;

Figure 15 is a PV diagram when a heavy load is on the engine of the second embodiment;

Figure 16 is a transverse cross sectional view of the cylinder head portion of a Diesel engine according to a third embodiment of the present invention;

Figure 17 is a longitudinal sectional view of the engine in Figure 16:

Figure 18 is a graph showing the relationship between the movement of the piston and the opening areas of the intake and exhaust valves when a heavy load is on the engine of the third embodiment;

Figure 19 is a graph showing the relationship between the movement of the piston and the opening areas of the intake and exhaust valves when the engine is started and when a light load is on the engine of the third embodiment;

Figure 20A and Figure 20B are PV diagrams illustrated by comparing the PV diagrams relating to the third embodiment, and Figure 20A shows the PV diagram when the engine is started and under a light load and Figure 20B shows the PV diagram under a heavy load;

Figure 21 is a transverse cross sectional view of the cylinder head portion of a gasoline engine relating to a fourth embodiment of the present invention;

Figure 22 is a longitudinal sectional view of the engine in Figure 21;

Figure 23 is a graph showing the relationship between the movement of the piston and the opening areas of the intake and exhaust valves when the engine is started and a light load is on the engine of the fourth embodiment;

Figure 24 is a graph showing the relationship between the movement of the piston and the opening areas of the intake and exhaust valves when a heavy load is on the engine of the fourth embodiment;

Figure 25 is an elevational view showing a gear train of a variable valve timing device relating to a fifth embodiment of the present invention;

Figure 26 is a sectional view along line Z to Z of Figure 25;

Figure 27 is a graph showing an average axial effective pressure and so on at a specified compression ratio of a conventional engine;

Figure 28 is a general view of a conventional early closing Miller cycle engine;

Figure 29A shows the relationship between the movement of the piston and the opening areas of the intake and exhaust valves in the engine in Figure 28 with a light load; and

Figure 29B shows the relationship between the movement of the piston and the opening areas of the intake and exhaust valves in the engine in Figure 28 with a heavy load.

Preferred embodiments of the variable compression ratio engine relating to the present invention will be described below in detail with reference to the attached drawings.

Figures 1 to 3 show a Diesel engine provided with an exhaust gas recirculating device relating to the first embodiment, and each cylinder has two intake valves and two exhaust valves. Reference number 25 indicates a piston, 26 and 27 are intake passages, and 28 is an exhaust passage. At a cylinder head 1, a first intake valve 2, a second intake valve 3, a first exhaust valve 4, a second exhaust valve 5, a first cam shaft 10, and a second cam shaft 20 are placed. On the first cam shaft 10, cams 11, 12, and 13 for the first intake valve 2, the first exhaust valve 4, and the second exhaust valve 5 are provided, and the cam 12 directly operates the first exhaust valve 4. The cams 11 and 13 respectively operate the first intake valve 2 and the second exhaust valve 5 by means of rocker arms 14 and 15.

On the second cam shaft 20, cams 21 and 22 are provided. The cam 21 directly operates the second intake valve 3, while the cam 22 operates the rocker arm 15 by oscillating a lever 23 placed at the cylinder head 1 with a pin 24 as its centre, and opens and closes the second exhaust valve 5. The second cam shaft 20 is rotated to a degree of an angle previously specified by a driving device which is not illustrated in the drawings, and can change the phases of the cams 21 and 22. Thereby the valve timing of the second intake valve 3 and the second exhaust valve 5 can be delayed.

The operation by the aforementioned structure will be described.

In Figure 4, the abscissa shows the position of the piston 25, the solid line shows the opening area of one valve, the two-dot chain line shows the total opening areas of two valves. A1 shows the exhaust valve, B1 shows the intake valve, and C1 shows the second exhaust valve. The first and second exhaust valves 4 and 5 start to open before the piston is at the bottom dead centre and close when the piston is in the vicinity of

the top dead centre, and always have the same phases. The first and second intake valves 2 and 3 start to open when the piston is in the vicinity of the top dead centre and close when the piston is at the vicinity of 90° before the bottom dead centre, and have the same phases. Then the second intake valve 3 opens when the piston is in the vicinity of the top dead centre and at the same time the second exhaust valve 5 opens for a short time as C1 shows. However, since the second exhaust valve 5 opens when the piston is in the vicinity of the top dead centre, most of the exhaust gas does not recirculate in the intake gas, so that there is no possibility that the fuel consumption increases or that the output is reduced.

Figure 5 shows the state under a heavy load, and in the intake stroke, intake is started at P0, and at P1a the first and the second intake valves 2 and 3 close, so that the pressure within the cylinder decreases and goes along the arrow to P1b. In the compression stroke, the pressure within the cylinder reaches P2a via P1b and P1a, in the combustion and expansion strokes, the pressure goes to P3 from P2a and then to P4, and in the exhaust stroke, the pressure goes to P1c from P4 and then to P0. In short, the engine operates in an early closing Miller cycle. At almost the end of the intake stroke, only the expansion and compression from P1a to P1b to P1a are conducted, so that the actual compression ratio is reduced to be as low as 11:1 to 13:1. Accordingly, high power output can be achieved.

On the other hand, when the load is light, as in Figure 1, the phases of the cams 21 and 22 are changed by rotating the second cam shaft 20 by the driving device and the closing time of the second intake valve 3 is set to be as late as the time when the piston is in the vicinity of the bottom dead centre. In Figure 6 showing the variation of the opening area under a light load, B11 shows the first intake valve 2, and B12 shows the second intake valve 3. Accordingly the second exhaust valve 5 is at the position shown in C1, that is. in the vicinity of 90° before the piston is at the bottom dead centre, and the exhaust gas is recirculated into the intake gas. As a result, the EGR rate becomes high. so that the generation of NOx is reduced.

Figure 7 is a PV diagram under a light load, and the engine operates in a normal cycle with the intake stroke from PO to P1, the compression stroke from P1 to P2, the combustion stroke from P2 to P3, the expansion stroke from P3 to P4, and the exhaust stroke from P4 to P1 to P0. The compression ratio in this cycle is from 15:1 to 17:1, and an excellent start and combustion state can be obtained.

The relationship between the aforementioned load on the engine and the compression ratio or the EGR rate will be described. In Figure 8, the most external curve is a torque curve when the engine operates at maximum power output. As shown in the drawing, when the load becomes heavier, the compression ratio becomes lower, in other words, when the load is lighter, the compression ratio becomes higher. As shown in Figure 9, when the load on the engine is lighter, the EGR rate becomes higher.

Then the second embodiment of the variable compression ratio engine relating to the present invention will now be described in detail with reference to the drawings.

Figures 10 to 12 show the essential parts of a gasoline engine provided with two intake and two exhaust valves per cylinder. Reference number 55 indicates a piston, 56 and 57 are intake passages, and 58 is an exhaust passage. On a cylinder head 31, a first intake valve 32, a second intake valve 33, a first exhaust valve 34, a second exhaust valve 35, a first cam shaft 40, and a second cam shaft 50 are placed. On the first cam shaft 40, cams 41, 42, and 43 for the first intake valve 32 and the first and second exhaust valves 34 and 35 are provided. The cam 41 operates the first intake valve 32 by means of a rocker arm 44, and the cam 42 directly operates the first exhaust valve 34. The cam 43 operates the second exhaust valve 35 by means of a lever 46 attached at the cylinder head 31 by a pin 45 so as to be free to oscillate.

On a second cam shaft 50, cams 51 and 52 are provided, and the cam 51 directly operates the second intake valve 33. The cam 52 oscillates the lever 46 by means of a lever 54 attached at the cylinder head 31 by a pin 53 so as to be free to oscillate, and opens and closes the second exhaust valve 35. The second cam shaft 50 is rotated at a previously specified angle relative to the shaft 40 by a driving device

which is not illustrated in the drawing, and by changing the phases of the cams 51 and 52, the valve timing of the second intake valve 33 and the second exhaust valve 35 can be delayed.

The operation by the aforementioned structure will be described. Figure 13 shows the variation of the opening area when a load is light, and the abscissa shows the position of the piston 55, the solid line shows the opening area of one valve, and the two-dot chain line shows the total opening areas of two valves. A2 corresponds to the two exhaust valves, B2 corresponds to the intake valves, and C2 corresponds to the second exhaust valve. The first and second exhaust valves 34 and 35 start to open before the piston is at the bottom dead centre, close when the piston is in the vicinity of the top dead centre, and have the same phase. On the other hand the first and second intake valves 32 and 33 have the same phases, and start to open when the piston is in the vicinity of the top dead centre and close when the piston is in the vicinity of the bottom dead centre. Then the second exhaust valve 35 opens for a short time in the vicinity of 90° before the piston is at the bottom dead centre, and the exhaust gas is recirculated into the intake gas. Accordingly the EGR rate increases and generation of NOx is reduced.

The cycle operation from the intake stroke to the exhaust stroke under a light load has the same basic cycle pattern as that in Figure 7 of the first embodiment. The compression ratio in this cycle operation is in the range of 11:1 to 13:1, so that the starting efficiency and thermal efficiency are improved, and the fuel consumption and generation of CO_2 can be reduced.

Figure 14 shows the variation of the opening area under a heavy load, and A2 corresponds to the exhaust valve, B21 corresponds to the first intake valve 32, B22 corresponds to the second intake valve 33, and C2 corresponds to the second exhaust valve 35. Under the aforementioned heavy load, the second cam shaft 50 is rotated by a driving device which is not illustrated in the drawing, and the second intake valve 33 closes at the position of 90° after the piston is at the bottom dead centre. Accordingly, the second exhaust valve 35 opens and closes when the piston is in the vicinity of the bottom dead centre, so that almost all the

exhaust gas does not recirculate into the intake gas. Therefore an increase in fuel consumption and reduction of output can be prevented.

Figure 15 is a PV diagram under a heavy load, and an intake of gas is conducted in the intake stroke from P0 to P1. In the compression stroke, the pressure does not increase from P1 to P1d since the second intake valve 33 is open, and since the second intake valve 33 closes at the point P1d, the pressure increases from P1d to P2b. This stroke is followed by the combustion stroke from P2b to P3, the expansion stroke from P3 to P4, and the exhaust stroke from P4 to P1 to P0, and the engine operates in a late closing Miller cycle. The compression ratio at this time is 8:1 to 10;1, so that high power output can be achieved and the occurrence of knocking under a high output is prevented.

The third embodiment of the variable compression ratio engine relating to the present invention will now be described in detail with reference to the drawings.

Figure 16 and Figure 17 show a Diesel engine having two intake valves and two exhaust valves for each cylinder, and at a cylinder head 101, a first intake valve 102, a second intake valve 103, a first exhaust valve 104, a second exhaust valve 105, a first cam shaft 110, and a second cam shaft 120 are placed. On the first cam shaft 110, cams 111, 112, and 113 for the first intake valve 102, the first exhaust valve 104, and second exhaust valve 105 are provided. The cam 112 directly operates the first exhaust valve 104, and the cams 111 and 113 operate the first intake valve 102 and the second exhaust valve 105 by means of rocker arms 114 and 115.

On a second cam shaft 120, a cam 121 is provided and directly operates the second intake valve 103. The second cam shaft 120 is rotated at a previously specified angle relative to cam shaft 110 by a driving device which is not illustrated in the drawings, and the valve timing of the second intake valve 103 can be delayed by changing the phase of the cam 121. Reference number 122 indicates a piston, 123 and 124 are intake passages. and 125 is an exhaust passage.

The operation by the aforementioned structure will be described.

Figure 18 shows the variation of the opening area under a heavy load, wherein the abscissa is the position of the piston 122, the solid line is an opening area of one valve, the two-dot chain line is the total opening areas of two valves. A3 corresponds to the exhaust valve, and B3 corresponds to the intake valve. The first and second exhaust valves 104 and 105 begin to open before the piston is at the bottom dead centre and close when the piston is in the vicinity of the top dead centre. Their phases are always the same. The first and the second intake valves 102 and 103, which have the same phases, begin to open when the piston is in the vicinity of the top dead centre and close in the vicinity of 20° to 90° before the piston is at the bottom dead centre.

The cycle operation from the intake stroke to the exhaust stroke under a heavy load has the same basic cycle pattern as that in Figure 5 of the first embodiment as shown in Figure 20B. Accordingly the engine operates in an early closing Miller cycle like in the first embodiment under a heavy load, and the actual compression ratio is as low as 11:1 to 13:1, so that high power can be outputted.

On the other hand, at starting and under a light load, the phase of the cam 121 is changed by rotating the second cam shaft 120 relative to the camshaft 110 by the driving device and the time when the second intake valve 103 closes is delayed to the time when the piston is in the vicinity of the bottom dead centre. In Figure 19 showing the variation of the opening area in this case, B31 corresponds to the first intake valve 102, and B32 corresponds to the second intake valve 103. Accordingly, the intake valves open when the piston is in the vicinity of the top dead centre and closes when the piston is in the vicinity of the bottom dead centre. From the intake stroke to the exhaust stroke, the basic cycle pattern becomes a normal cycle operation the same as in Figure 7 of the first embodiment and the compression ratio is 15:1 to 17:1.

The difference between the aforementioned state under a heavy load and the state at the starting time or under a light load will be described with reference to Figure 20A and Figure 20B. The compression ratio under a heavy load is as small as 11:1 to 13:1, so that the compression pressure P2a is lower than P2 and there is a margin up to P3

which is a maximum allowable pressure Pmax of the engine, therefore a lot of fuel can be combusted. As a result, the area surrounded by P1c to P1a to P2a to P3 to P4 under a heavy load is larger than the area surrounded by P1 to P2 to P3 to P4 at the starting time and so on. Accordingly the amount of work done under a heavy load is large and high power output is maintained, so that an engine which is small in size but has a high power output can be realized. Moreover, the intake part has no excessive volume, so that the engine can operate in an efficient Miller cycle. On the other hand, at starting and under a light load, the compression ratio is as large as 15:1 to 17:1, so that an excellent start and combustion state can be obtained.

Next, the fourth embodiment of the variable compression ratio engine relating to the present invention will be described in detail with reference to the drawings.

Figure 21 and Figure 22 show a gasoline engine provided with two intake and two exhaust valves for each cylinder, and at a cylinder head 131, a first intake valve 132, a second intake valve 133, a first exhaust valve 134, a second exhaust valve 135, a first cam shaft 140, and a second cam shaft 150 are placed. On the first cam shaft 140, cams 141, 142, and 143 for the first intake valve 132, the first exhaust valve 134, and second exhaust valve 135 are provided. The cam 141 operates the first intake valve 132 by means of a rocker arm 144, and the cams 142 and 143 directly operate the first exhaust valve 134 and the second exhaust valve 135, respectively.

On the second cam shaft 150, a cam 151 is provided and directly operates the second intake valve 133. The second cam shaft 150 is rotated at a previously specified angle relative to cam shaft 140 by a driving device which is not illustrated in the drawings, and can delay the valve timing of the second intake valve 133 by changing the phase of the cam 151 relative to the other cams. Reference number 152 indicates a piston, 153 and 154 are intake passages, and 155 is an exhaust passage.

The operation in the aforementioned structure will be described. In Figure 23, the solid line is an opening area of one valve, and the fine two-dot chain line shows the total opening area of two valves, with A4

corresponding to the exhaust valve and B4 corresponding to the intake valve. The first and second exhaust valves 134 and 135 begin to open before the piston is at the bottom dead centre, close when the piston is in the vicinity of the top dead centre, and always have the same phase. The first intake valve 132 and the second intake valve 133 have the same phase, and begin to open when the piston is in the vicinity of the top dead centre, then close when the piston is in the vicinity of the bottom dead centre.

From the intake stroke to the exhaust stroke at starting and under a light load, the basic cycle pattern is the same normal cycle operation as in Figure 7 of the first embodiment, and the compression ratio is 11:1 to 13:1. Accordingly, as in the second embodiment, the starting efficiency and thermal efficiency are improved, and the fuel consumption and the generation of CO_2 can be reduced.

Figure 24 shows the variation of the opening area under a heavy load, where B41 corresponds to the first intake valve 132, and B42 corresponds to the second intake valve 133. Under heavy load, the second cam shaft 150 is rotated relative to camshaft 140 by a driving device which is not illustrated in the drawings, and the second intake valve 133 closes at 40° to 90° after the piston is at the bottom dead centre.

From the intake stroke to the exhaust stroke under a heavy load, the basic cycle pattern is the same as in Figure 15 of the second embodiment, and the engine operates in a late closing Miller cycle. In the case of Figure 15, in the compression stroke from P1 to P1d, the pressure in the cylinder does not increase since the second intake valve 133 is open, and only increases from P1d to P2b because the second intake valve 133 closes at the point P1d. The compression ratio at this time is 8:1 to 10:1, and a high power is outputted and the occurrence of knocking is prevented as in the second embodiment. In addition, the intake part has no excessive volume, so that an efficient variable compression ratio engine can be obtained.

Next, the fifth embodiment of the variable compression ratio engine relating to the present invention will be described in detail with reference to the drawings.

The variable compression ratio engine of the present embodiment is a double overhead camshaft type of engine which can convert between a normal cycle and a Miller cycle, and has the engine in the third embodiment as a base with the variable valve timing device being provided. In Figure 16, the second cam shaft 120 can be rotated as shown by the arrows by the variable valve timing device described below. By this device, the valve timing can be changed over a range of 70° to 90° in relation to the crankshaft angle, so that the actual compression ratio is variable and high power can be outputted.

Figure 25 and Figure 26 show a gear train placed at the end portions of the first and second cam shafts 110 and 120 in Figure 16. A sun gear 230 is fixedly attached at the first cam shaft 110, and a ring gear 231 is attached at the foremost end of the first cam shaft 110 by means of a bearing 232 so as to be free to rotate. The ring gear 231 is provided with internal gear teeth 233 and external gear teeth 234. A set of three planet gears 235 mesh with the sun gear 230 and with the internal gear teeth 233 of ring gear 231. A support shaft 236 of each planet gear 235 is fixedly attached to a planet carrier 237. The planet carrier 237 is attached at the cylinder head 101 by a shaft so as to freely rotate, and a worm wheel sector 238 is provided at the outer perimeter of the planet carrier. The worm wheel sector 238 is meshed with a worm 241 driven by an electric motor 240.

On the second cam shaft 120 supported by a shaft at the cylinder head 101, a gear 242 meshes with the external gear teeth 234 of ring gear 231, and a timing gear 243 is also fixedly attached. The timing gear 243 is meshed with an idler gear 244, which in turn meshes with a crank gear 246 fixedly attached to the engine crankshaft 245.

Here, when the number of teeth of the sun gear 230 is Z1, the number of internal gear teeth 233 of the ring gear 231 is Z2, the number of teeth of the gear 242 is Z3, and the number of external gear teeth 234 of the ring gear 231 is Z4, then $Z1 / Z2 = Z3 / Z4$. The ratio of the numbers of teeth of the crank gear 246 and the timing gear 243 is 1 : 2. Accordingly, the rotational speed of the second cam shaft 120 is half of the rotational speed of the crankshaft 245, and the second cam shaft 120 and the first cam shaft 110 have the same rotational speed.

When the phase of the first cam shaft 110 and the second cam shaft 120 is to be changed, the worm 241 is rotated by the electric motor 240, and the worm wheel sector 238 is rotated by a specified angle. The planet carrier 237 rotates at the same time since the carrier 237 is integrated with the worm wheel sector 238, and the planet gear 235 rotates the ring gear 231 by revolving around the sun gear 230 as the planet gear 235 is rotating. Accordingly, the gear 242 rotates, and the phase of the second cam shaft 120 to the first cam shaft 110 is changed. The rotational angle ratio τ of the gear 242 and the worm wheel 238 at this time can be obtained from the following formula:

$$\tau = [(Z1 + Z2) / Z2 1 Z4 / Z3$$

Accordingly, the variable valve timing device, in which the gear 242 rotates through a large angle relative to sun gear 230 by only rotating the worm wheel sector 238 through a small angle, can easily give a phase difference of 70° to 90° between the first cam shaft 110 and the second cam shaft 120.

Summing up the aforementioned present embodiments, two cam shafts are connected by the medium of a planetary gear unit consisting of the sun gear, ring gear and the planet gear, and one cam shaft is fixedly attached to the sun gear while the other cam shaft is fixedly attached to the gear meshed with the ring gear. The carrier supporting the planet gear is rotatably attached to a case supporting the sun gear shaft, and is connected to a rotational driving device. Accordingly, the rotational speed of the gear meshed with the ring gear is increased, relative to the rotation of the carrier. In other words, when the carrier is rotated over a small angle by the rotational driving device, the gear is rotated over a large angle.

The present invention is useful as a variable compression ratio engine which can convert between an early closing or a late closing Miller cycle operation and a normal cycle operation and which can reduce the generation of NOx and so on and can prevent the occurrence of knocking.

CLAIMS:

1. A variable compression ratio engine which is provided with two or more intake valves and one or more exhaust valves per cylinder and which changes the compression ratio by opening and closing said intake valves and/or said exhaust valves by cams provided on two or more cam shafts, comprising:

an exhaust gas recirculating device including a first cam shaft provided with the cams for operating the intake valve and said exhaust valves and a second cam shaft provided with the cams for operating at least one of said intake valves and the exhaust valve; and

a valve driving device whereby, in the intake stroke at a specified driving time, the closing time of said intake valves is set at a time before a piston is at the bottom dead centre with the opening and closing times of the exhaust valve being at a time when the piston is in the vicinity of the top dead centre, and whereby, in the intake stroke under a light load, the closing time of at least one of said intake valves is set at a time when the piston is in the vicinity of the bottom dead centre with the opening and closing times of the exhaust valve being set at a time before the piston is at the bottom dead centre, said valve driving device recirculating part of the exhaust gas into intake gas under the aforementioned light load by changing the phase of the cams on said second cam shaft.

2. A variable compression ratio engine which is provided with two or more intake valves and one or more exhaust valves per cylinder and which changes the compression ratio by opening and closing said intake valves and/or said exhaust valves by the cams provided on two or more cam shafts, comprising:

the exhaust gas recirculating device including the first cam shaft provided with the cams for operating the intake valve and said exhaust valves and the second cam shaft provided with the cams for operating at least one of said intake valves and the exhaust valve; and

the valve driving device whereby, in the intake stroke under a light load, the closing time of said intake valves is set at a time when the piston is in the vicinity of the bottom dead centre with the opening and closing times of the exhaust valve being set at a time before the piston is at the bottom dead centre, and whereby, in the intake stroke at a specified driving time, the closing time of at least one of said intake valves is set at a time after the piston is at the bottom dead centre with the opening and closing times of the exhaust valve being set at a time when the piston is in the vicinity of the bottom dead centre, said valve driving device recirculating part of the exhaust gas into intake gas under the aforementioned light load by changing the phase of the cams on said second cam shaft.

3. A variable compression ratio engine which is provided with two or more intake valves and one or more exhaust valves per cylinder and which changes the compression ratio by opening and closing said intake valves and/or said exhaust valves by the cams provided on two or more cam shafts, comprising:

an intake device which changes the valve timing of at least one of said intake valves by changing the phase of the cam for opening and closing the intake valve, said intake device setting the closing time of said intake valves at a time before the piston is at the bottom dead centre, and said intake device, at a specified driving time, setting the closing time of at least one of said intake valves at a time when the piston is in the vicinity of the bottom dead centre.

4. A variable compression ratio engine according to Claim 6, wherein the aforementioned closing time of the intake valves which is set at a time before the piston is at the bottom dead centre is a time when a crank rotational angle is at 20° to 90° before the piston is at the bottom dead centre.

5. A variable compression ratio engine which is provided with two or more intake valves and one or more exhaust valves per cylinder and which changes the compression ratio by opening and closing said intake valves and/or said exhaust valves by the cams provided on two or more cam shafts, comprising:

the exhaust gas recirculating device including the first cam shaft provided with the cams for operating the intake valve and said exhaust valves and the second cam shaft provided with the cams for operating at least one of said intake valves and the exhaust valve;

a planetary gear unit provided with a sun gear fixedly attached at said first cam shaft, a ring gear, a gear meshed with said ring gear and fixedly attached at said second cam shaft, and a planet gear; and

a variable valve timing device which changes the valve timing by adjusting the phase of said first and second cam shafts by freely changing the relative positional relationship between the support shaft of said planet gear and the shaft of said sun gear, said variable valve timing device being operated so that part of the exhaust gas is recirculated into intake gas.

6. A variable compression ratio engine substantially as herein described with reference to Figures 1 to 3, Figures 10 to 12, Figures 16 to 18, figures 21 and 22, or Figures 25 and 26 of the accompanying drawings.

INTERNATIONAL SEARCH REPORT

International application No. PCT/JP95/00341

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl^6 F02D13/02, F02M25/07, F01L1/26, F01L1/34, F01L13/00

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

Int. Cl^6 F02D13/02, F02M25/07, F01L1/26, F01L1/34, F01L13/00

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Jitsuyo Shinan Koho	1926 - 1995
Kokai Jitsuyo Shinan Koho	1971 - 1995

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP, 53-40115, A (Hino Motors, Ltd.), April 12, 1978 (12. 04. 78), Lines 9 to 14, upper left column, page 2 (Family: none)	1, 2
Y	JP, 3-55643, B2 (Mazda Motor Corp.), August 26, 1991 (26. 08. 91) (Family: none)	1, 2
Y	JP, 54-129219, A (Isao Matsui), October 6, 1979 (06. 10. 79), Fig. 2 (Family: none)	1, 2
A	JP, 53-40115, A (Hino Motors, Ltd.), April 12, 1978 (12. 04. 78) (Family: none)	3, 4
A	JP, 3-55643, B2 (Mazda Motor Corp.), August 26, 1991 (26. 08. 91) (Family: none)	3, 4
A	JP, 54-129219, A (Isao Matsui), October 6, 1979 (06. 10. 79) (Family: none)	3, 4

[X] Further documents are listed in the continuation of Box C. [ ] See patent family annex.

* Special categories of cited documents:
- "A" document defining the general state of the art which is not considered to be of particular relevance
- "E" earlier document but published on or after the international filing date
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- "O" document referring to an oral disclosure, use, exhibition or other means
- "P" document published prior to the international filing date but later than the priority date claimed
- "T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
- "X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
- "Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
- "&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
May 18, 1995 (18. 05. 95)	June 13, 1995 (13. 06. 95)

Name and mailing address of the ISA/	Authorized officer
Japanese Patent Office	
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C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP, 57-51536, B2 (Kanesaka Gijutsu Kenkyusho K.K.), November 2, 1982 (02. 11. 82) (Family: none)	3
X	JP, 5-80561, B2 (Mazda Motor Corp.), November 9, 1993 (09. 11. 93), Fig. 3 (Family: none)	5, 8, 9
A	JP, 5-80561, B2 (Mazda Motor Corp.), November 9, 1993 (09. 11. 93) (Family: none)	6, 7
A	JP, 57-51536, B2 (Kanesaka Gijutsu Kenkyusho K.K.), November 2, 1982 (02. 11. 82) (Family: none)	6, 7
X	JP, 2-135604, U (Mitsubishi Motors Corp.), November 13, 1990 (13. 11. 90) (Family: none)	10 - 12

国際調査報告

国際出願番号 PCT/JP 95/00341

A. 発明の属する分野の分類（国際特許分類（IPC））

Int. Cl⁶ F02D13/02, F02M25/07, F01L1/26, F01L1/34, F01L13/00

B. 調査を行った分野

調査を行った最小限資料（国際特許分類（IPC））

Int. Cl⁶ F02D13/02, F02M25/07, F01L1/26, F01L1/34, F01L13/00

最小限資料以外の資料で調査を行った分野に含まれるもの

日本国実用新案公報 1926－1995年
日本国公開実用新案公報 1971－1995年

国際調査で使用した電子データベース（データベースの名称、調査に使用した用語）

C. 関連すると認められる文献

引用文献のカテゴリー＊	引用文献名 及び一部の箇所が関連するときは、その関連する箇所の表示	関連する請求の範囲の番号
Y	JP, 53－40115, A（日野自動車工業株式会社），12. 4月. 1978（12. 04. 78），第2頁左上欄，第9－14行（ファミリーなし）	1, 2
Y	JP, 3－55643, B2（マツダ株式会社），26. 8月. 1991（26. 08. 91）（ファミリーなし）	1, 2
Y	JP, 54－129219, A（松井 功），6. 10月. 1979（06. 10. 79），	1, 2

☑ C欄の続きにも文献が列挙されている。

☐ パテントファミリーに関する別紙を参照。

＊ 引用文献のカテゴリー
「A」特に関連のある文献ではなく、一般的技術水準を示すもの
「E」先行文献ではあるが、国際出願日以後に公表されたもの
「L」優先権主張に疑義を提起する文献又は他の文献の発行日若しくは他の特別な理由を確立するために引用する文献（理由を付す）
「O」口頭による開示、使用、展示等に言及する文献
「P」国際出願日前で、かつ優先権の主張の基礎となる出願の日の後に公表された文献

「T」国際出願日又は優先日後に公表された文献であって出願と矛盾するものではなく、発明の原理又は理論の理解のために引用するもの
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「Y」特に関連のある文献であって、当該文献と他の1以上の文献との、当業者にとって自明である組合せによって進歩性がないと考えられるもの
「&」同一パテントファミリー文献

国際調査を完了した日 18. 05. 95	国際調査報告の発送日 13.06.95
名称及びあて先 日本国特許庁（ISA/JP） 郵便番号100 東京都千代田区霞が関三丁目4番3号	特許庁審査官（権限のある職員） 3G 7536 宮崎侑久 印 電話番号 03-3581-1101 内線 3355

様式PCT/ISA/210（第2ページ）（1992年7月）